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FEASIBILITY STUDY ON
THE USE OF PLASTIC COMPONENTS FOR
THE 81 MM MORTAR BIPOD ASSEMBLY

MERRILL EIG

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FEASIBILITY STUDY ON
THE USE OF PLASTIC COMPONENTS
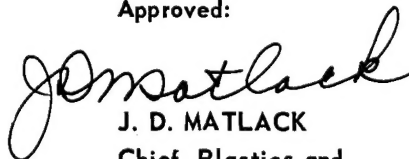# FOR THE 81 MM MORTAR BIPOD ASSEMBLY

by

Merrill Eig

February 1963

Feltman Research Laboratories
Picatinny Arsenal
Dover, N. J.

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OBJECT

To analyze strain gage data on the bipod assembly for the 81 mm mortar
and to determine the feasibility of replacing heavy metal parts with light-
weight plastics.

SUMMARY

Strain data obtained during firing from gages positioned on the legs of the
81 mm mortar bipod assembly was reported to Picatinny Arsenal by
Watervliet Arsenal. The forces exerted on the bipod assembly were deter-
mined by analyzing this data. Once this was accomplished the individual
forces acting upon components could be calculated. A 45° elevation was
used, with the bipod traverse assembly located in the central position.

The calculated maximum stresses were as follows: a bearing stress of
20,100 psi on the case; and a 2,780 psi stress due to bending and a com-
pressive stress of 22,880 psi for the tube housing. The cover, body, and
yoke ring were not significantly stressed. The connector showed a bending
stress of 17,900 psi and a shear stress of 13,500 psi.

Rolled reinforced plastic tubing was suggested for fabrication of proto-
type components, as it can be easily machined to the desired shapes and
has the required strength. It should be noted that the design strength of the
tubing is marginal.

For end items, a glass-filled epoxy compression molding compound such
as Scotchply 1100 would have the desired strength and other required
physical properties.

The components were redesigned for fabrication from plastics. The
recommended design changes are shown in an Appendix.

CONCLUSIONS

It may be concluded from this analysis that it is feasible to fabricate
the desired components of the 81 mm mortar bipod assembly from plastic
materials.

RECOMMENDATIONS

It is recommended that prototype components be fabricated and tested under actual firing conditions to determine suitability of design. It is recommended that, upon acceptance of design, molds be fabricated and a glass filled epoxy-molding material such as Scotchply 1100 be used for production of end items requiring high strength.

INTRODUCTION

The Plastics and Packaging Laboratory at Picatinny was requested by Watervliet Arsenal to analyze strain gage data obtained by Watervliet Arsenal on the bipod assembly for the 81 mm mortar and from such data, to determine the feasibility of replacing existing metal components of the mortar assembly with lightweight plastic parts. The use of high strength, lightweight plastics for this application was considered desirable as a means of lightening and making more portable an assembly designed for use and handling by infantry soldiers.

DISCUSSION

The strain gage data received from Watervliet Arsenal was obtained during actual firings of the mortar. Rectangular strain rosettes consisting of three gages laid out at 45° and 90° angles to each other were used to measure strain on the bipod legs.

Two C-7 type strain gages in series 180° apart were used to measure strain on the neck of the base plate.

It was apparent, from data on the first 10 rounds, that the forces acting on the bipod legs are almost solely axial; i.e., tend to extend or compress the legs. In such a situation, the rosette gage is not needed. Therefore, on the remaining fifteen rounds, only the strain recorded by the axial gage was used. Maximum positive and negative deflections were read and the stress values were computed as the product of strain and Young's Modulus (see Table 3, p 14).

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Review of this reported strain gage data by Picatinny Arsenal indicated that the number of gages placed around the bipod legs was not sufficient for determining the actual maximum stress. Therefore, an engineering approach utilizing an iteration process was tried and successfully used (see Appendix A) to determine the maximum stress.

Determination of Tensile Bending (σ_{TB}) and Compressive Stresses (σ_C) in Right Leg

From the firing data reported to Picatinny Arsenal by Watervliet Arsenal, it was observed that the right leg of the bipod experienced a compressive load of 6670 psi after 11.4 milliseconds after firing (see Table 3). After 40.4 milliseconds, a stress reversal occurred in which this compressive stress changed to a tensile stress of 17,980 psi. It is important to note that the stresses reported in the data and substantiated by Watervliet Arsenal are the sums or resultants of the tensile bending stress (σ_{TB}) and the compressive stress (σ_C). It must therefore be assumed that there exists a tensile stress (due to bending) such that, when a compressive stress of 6,670 psi is added to it, the resultant is a tensile stress of 17,980 psi. The reason is that both tensile and compressive stresses can conceivably be applied simultaneously. Thus, by numerically adding the compressive and tensile stresses reported in the data, values for σ_{TB} and σ_C are determined (see Appendix A). For this particular example, σ_{TB} is determined to be 24,650 psi and σ_C is 6,770 psi. This procedure is continued for each of the six test rounds to determine σ_{TB} and σ_C for each case. Use of the maximum values of σ_{TB} and σ_C will represent the maximum possible loading condition which can be sensed by the right bipod leg. The values thus determined are:

$$\sigma_{TB} = 24,650 \text{ psi}$$

$$\sigma_C = 12,760 \text{ psi}$$

It is important to re-emphasize that these stress values are resultant stresses; each being the sum of a tensile bending stress and a compressive stress. Also, that it is possible to obtain these values from an infinite number of tensile and compressive stresses. For example, suppose the leg assembly is subjected simultaneously to a tensile bending stress of 37,980 psi and a compressive stress of 20,000 psi. The resultant stress would be 17,980 psi. It is seen that this is exactly the value of stress reported by the strain gage data, yet it is by no means the maximum stress,

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as demonstrated, since the maximum recorded stress is a function of the time lag in applied loads. Similarly, a reverse procedure can be made to yield a negative value of stress, i.e., 6,770 psi.

In order to limit the many possible choices of bending and compressive stresses, it will be assumed, and reasonably so, that approximately 15% of the reactive force (or 12,000 lb) is transmitted to the bipod in the form of friction (F_f, see Fig 5). Having established this criterion, an iteration process will now be used to converge upon the final design stress values. This is accomplished by substituting various combinations of stresses the sum of which is a plus 17,980 psi into Equation 6 (Appendix A) until the desired value of F_f is obtained. When this condition is satisfied, the maximum design value for σ_{TB} is obtained. This same procedure is repeated for various values of stresses yielding a resultant stress of a minus 6,770 psi. Similarly, when these values are substituted into Equation 6 until the desired value of F_f is obtained, a maximum value of σ_C is also obtained.

It can be shown by this iteration process that there are two sets of values that will yield F_f equal to 12,000 pounds

$$\text{(a)} \qquad \sigma_{TB} = 33,000 \text{ psi}$$

$$\sigma_C = 16,000 \text{ psi}$$

$$\text{(b)} \qquad \sigma_{TB} = 3,000 \text{ psi}$$

$$\sigma_C = 15,760 \text{ psi}$$

Since the values in (a) represent the more extreme case, these values will be used for design purposes. It is interesting to note that they are in reasonably good agreement with the maximum values determined from the data; i.e., $\sigma_{TB} = 24,650$ psi and $\sigma_C = 12,760$ psi.

From Equation 6, it is obvious that σ_C is the major contributing factor since the coefficient of σ_{TB} is very small compared to that of σ_C. For all practical purposes, F_f is a function of σ_C only. This fact also agrees with the comments reported on the data, to the effect that strain rossettes are not needed since most of the stress is compressive.

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Determination of σ_{TB} and σ_C in Left Leg

The process for determining σ_{TB} and σ_C for the left leg is identical to to that described for the right leg. The only difference is in the physical dimensions. The applicable parameters are listed below.

Moment of inertia (I)	.0328 in.^{4}
Length of leg (l)	27 in.
Distance from neutral axis of leg to extreme fiber (c)	.476 in.
Area (A)	.404 in.^{2}

Substitution of these values into Equation 5 (see Appendix A) yields the following equation for frictional force:

$$F_f = \sqrt{(26.2 \times 10^{-6})\, \sigma_{TB}^2 + (.652)\, \sigma_C^2} \tag{1}$$

Using the iteration method, the maximum design stresses are determined to be

$$\sigma_{TB} = 44,000 \text{ psi}$$

$$\sigma_C = 17,000 \text{ psi}$$

Here again the stresses are in reasonably good agreement with those determined from the data

$$\sigma_{TB} = 27,260 \text{ psi}$$

$$\sigma_C = 11,890 \text{ psi}$$

Substituting the calculated values for σ_{TB} and σ_C into Equation 5 (in Appendix A) yields a fricitonal force for the right and left leg respectively,

$$F_{f_R} = 12,540 \text{ lb}$$

$$F_{f_L} = 13,560 \text{ lb}$$

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F_{favg} = 13,050 pounds. Note that this value does not depart significantly from the 15% (or 12,000 pounds) originally assumed. If the calculated value for σ_C for left and right leg is substituted into Equation 2d and 4b respectively (Appendix A) an F''_{favg} of 15,140 pounds is obtained.

At this point, all the forces acting on the system are known and a detailed stress analysis can be made of the various components.

Stresses Induced in Plastic Components

Hub Connection

It is contemplated that the yoke and the tube housing will be fastened to each other by bonding with an epoxy adhesive rather than by the use of screws. The applied load F''_{favg} will thus induce an interfacial shear stress (see Fig 1, pg 8).

$$\sigma_{sh} = \frac{F''_{favg}}{A} = \frac{15,140}{\pi \, Dh} = \frac{15,140}{\pi \, (1.6875)(1.75)} = 1,635 \, psi$$

Which is well below the design value of 3000 psi for epoxy resins.

Case

Upon firing, shaft A (Fig 2, p 9) imparts a bearing load to the case, with a resulting bearing stress of

$$\sigma_{BR} = \frac{F''_{favg}}{A} = \frac{15,140}{\frac{\pi}{4}(1.6895^2 - 1.375^2)} = 20,100 \, psi$$

Tube Housing

The force acting on the left leg, for a calculated stress of 44,000 psi is

$$R'_{f_L} = (F_{TB})_L = \frac{\sigma_B \, 2I}{cl} = \frac{44,000 \, (2)(.0328)}{(.476)(27)} = 224 \, lb$$

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Similarly, a stress of 33,000 psi will induce a force in the right leg of

$$R'_{f_R} = (F_{TB})_R = \frac{33,000 \, (2) \, (.0498)}{(.563) \, (27)} = 216 \text{ lb,}$$

the average force being

$$(F_{TB})_{avg} = \frac{(F_{TB})_L + (F_{TB})_R}{2} = 220 \text{ lb.}$$

The tube housing shaft B (Fig 1) will act as a solid shaft during firing since shafts A and B are coaxial. The moment of inertia is

$$I_H = \frac{\pi}{64} \, (D)^4 = \frac{\pi}{64} \, (1.6895)^4 = .402 \text{ in.}^4$$

The stress due to bending in the tube housing is

$$\sigma_B = \frac{Mc}{I} = (F_{TB})_{avg} \frac{lc}{I} = \frac{(220) \, 6}{.402} \left(\frac{1.6895}{2} \right)$$

$$\sigma_B = 2,780 \text{ psi}$$

The bearing stress of 20,100 psi imparted to the case is the same compressive stress as exists in the housing. Thus the total stress in the tube housing is the sum of the bending and compressive stresses,

$$\sigma_T = \sigma_B + \sigma_C = 2,780 + 20,100 = 22,880 \text{ psi.}$$

It is to be noted that the cover and body and the yoke ring components do not sense any significant stresses since the only applied load is the inertia of their own weight. However, the yoke ring must withstand a sustained shear load at temperatures of 400°- 500°F for periods of 15-30 minutes.
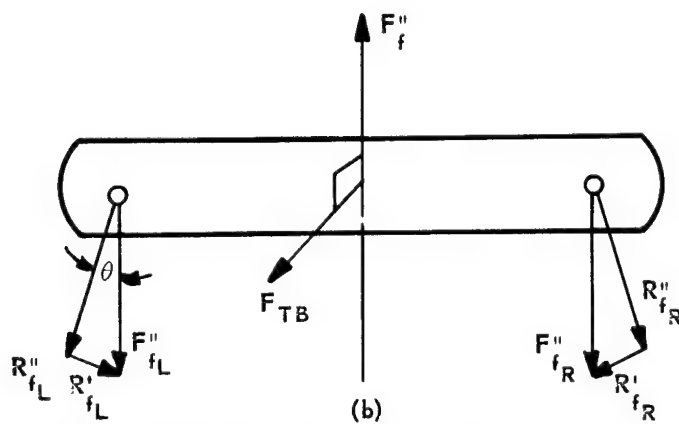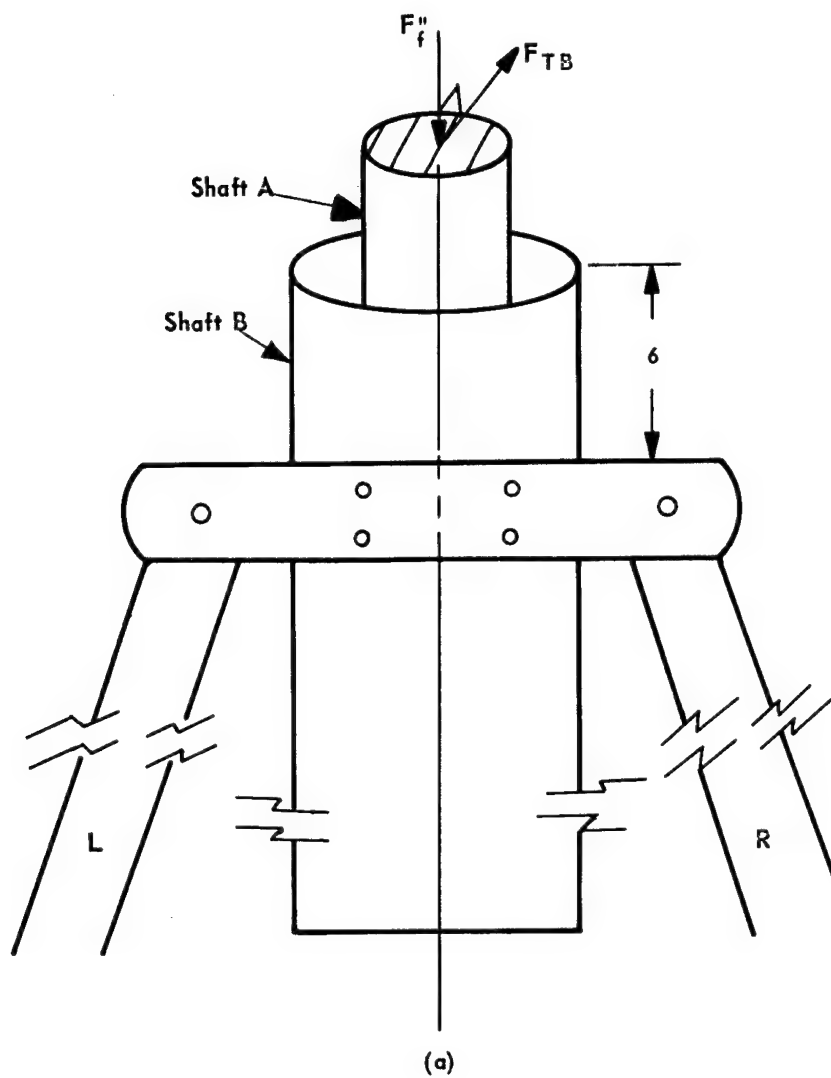
Fig 1   Bipod assembly (a) and yoke assembly (b)

$^{15}/_{16} + ^{1}/_{64}$

.9375

1.4375

1.4355

1.373

1.375

.6865

.6875
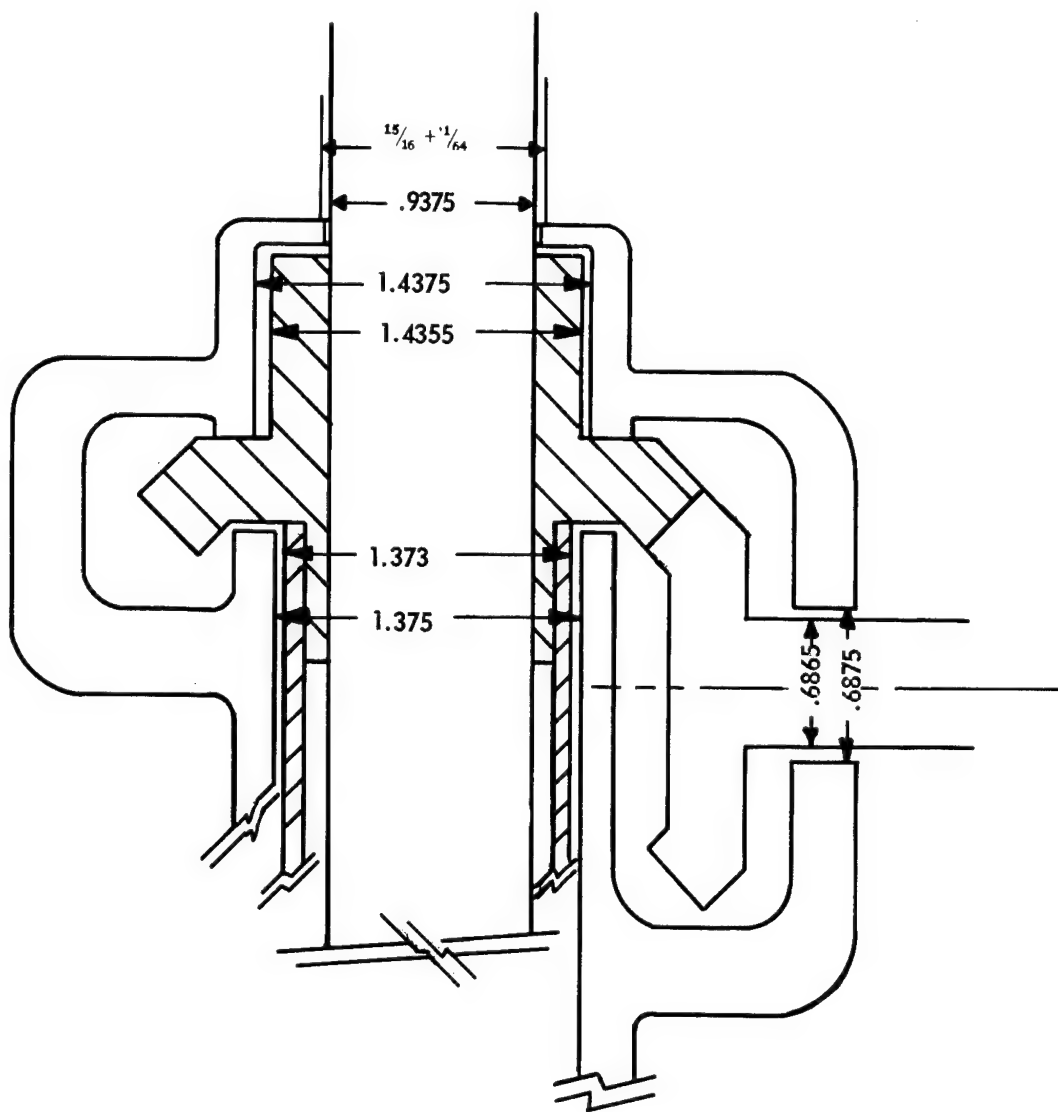
Fig 2   Schematic of elevating mechanism

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Connector

The maximum moment occurs at section AA (see Fig 3), with a moment of inertia equal to

$$I_{AA} \approx \frac{1}{12}hb^3 = \frac{1}{12}\left(1\frac{9}{16}\right)\left(1\frac{5}{16}\right)^3 = .297 \text{ in.}^4$$

and section BB has a smaller moment of inertia

$$I_{BB} \approx \frac{1}{12}(.312)\left(1\frac{5}{16}\right)^3 = .174 \text{ in.}^4$$

The corresponding bending moments are

$$M_{AA} = \left(\frac{F''_{favg}}{2}\right)l_1 = \frac{15,140}{2}(1.25) = 9,460 \text{ in.-lb}$$

$$M_{BB} = \frac{15,140}{2}(.72) = 5,440 \text{ in.-lb}$$

and the bending stresses are

$$\sigma_{AA} = \frac{Mc}{I} = \frac{9,460}{.297}\left(\frac{9}{16}\right) = 17,900 \text{ psi}$$

$$\sigma_{BB} = \frac{5,440}{.174}\left(\frac{9}{16}\right) = 17,650 \text{ psi}$$

The shear stress at section CC is,

$$\sigma_{sh} = \frac{F''_{favg}/2}{A} = \frac{\dfrac{15,140}{2}}{3[2\{(.312)(.3)\}]} = 13,500 \text{ psi}$$

The schematic diagram of the elevating mechanism (Fig 2, p 9) indicates the required dimensions of various mating parts. These dimensions were determined in order to preclude binding within the military specifications temperature range of $-65°F$ through $160°F$.

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Fig 3   Connector

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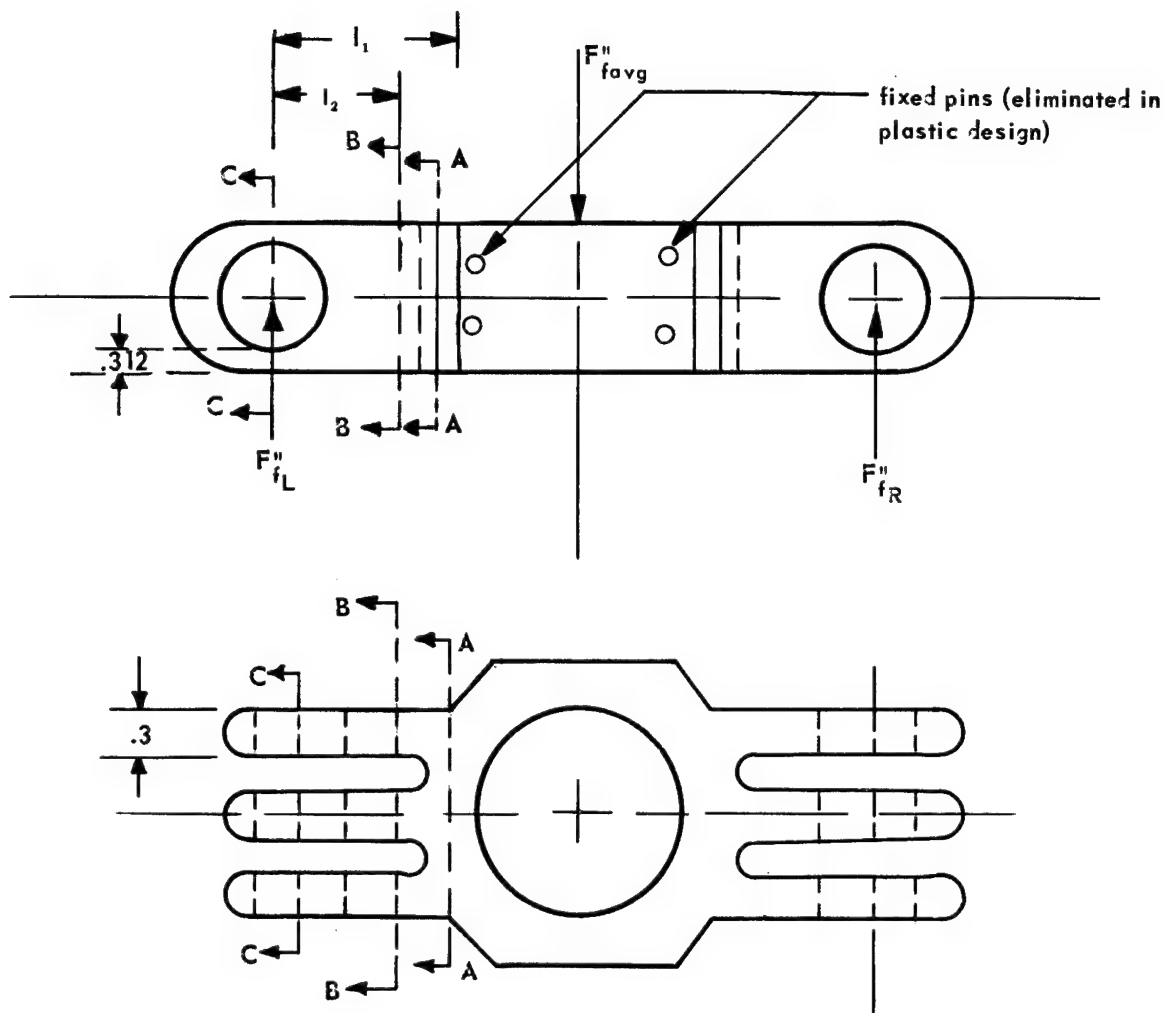
In terms of stress requirements mentioned above, the bipod assembly components in question can be fabricated from plastics. A molding mate- such as a fiberglass-filled epoxy should readily fulfill all of the require- ments for such plastic parts.

If fabrication of prototype parts is desired this may be accomplished by machining components from rolled tubing. The particular materials sug- gested for each prototype component are listed below in order of preference (1, 2, 3) on the basis of strength and thermal properties.

Comparison of design stress with the calculated stress values indicates that some of the strength requirements on the rolled tubing materials are marginal. During the analysis, however, the worst possible conditions were used in each case and it is recommended that materials with lower design stress values be test evaluated.

TABLE 1

Composition of component parts

	1		2		3	
	Reinforcement	Resin	Reinforcement	Resin	Reinforcement	Resin
Tube	Cotton cloth	Phenolic	Asbestos	Phenolic	Paper	Phenolic
Case	Cotton cloth	Phenolic	Asbestos	Phenolic	Paper	Phenolic
Bracket	Paper	Phenolic	Asbestos	Phenolic	Cotton cloth	Phenolic
Body	Paper	Phenolic	Asbestos	Phenolic	Cotton cloth	Phenolic
Cover	Paper	Phenolic	Asbestos	Phenolic	Cotton cloth	Phenolic
Connector	Glass	Silicone	Paper	Phenolic	Asbestos	Phenolic
Yoke ring	Glass	Silicone*	Glass	Silicone	Glass	Silicone

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*The silicone glass rolled tubing has the following physical properties:

Thermal coefficient of expansion $= 1.1 \times 10^{-5}$ cm/cm $^\circ$C

Maximum operating temperature, continuous $= 400^\circ$F

Maximum operating temperature, short time $= 475^\circ$F

The reported tensile and compressive strengths for the materials listed in Table 1 are as follows:

TABLE 2

Material design strength values

Materials		Tensile strength,	Compressive strength,
Reinforcement	Resin	psi	psi
Paper	Phenolic	10,500	17,500
Asbestos	Phenolic	8,500	19,000
Cotton	Phenolic	7,500	21,000
Glass	Silicone	30,000	15,000

The existing parts were redesigned for plastics (see Figs 10-16 in Appendix B, pp 25-31) taking into account the thermal coefficients of expansion of the recommended plastic materials.

Watervliet Arsenal did not request fabrication and testing of the proposed plastic parts. This report, therefore, is a feasibility and design criteria study only.

TABLE 3

Strain data submitted by Watervliet Arsenal

| Round Number | Bipod Traverse Position | Angle of Elevation, degrees | Gage Mounted Vertically on | | | | | |
| | | | Right Bipod Leg | | | Left Bipod Leg | | |
			Time,* ms	Strain, μ in./in.	Stress,** psi	Time,* ms	Strain, μ in./in.	Stress,** psi
323	Center	45	11.4	−230	−6,670	7.5	−410	−11,890
323	Center	45	40.4	620	17,980	44.6	530	15,370
324	Center	45	12.0	−210	−6,090	8.1	−230	−6,670
324	Center	45	40.2	270	7,830	28.3	320	−9,280
325	Max left	45	8.4	−340	−9,860	8.8	−390	−11,310
325	Max left	45	43.9	380	11,020	44.4	460	13,340
326	Max left	45	8.1	−310	−8,990	−	−	−
326	Max left	45	40.6	·600	17,400	−	−	−
327	Max right	45	9.2	−400	−11,600	−	−	−
327	Max right	45	36.4	350	10,150	−	−	−
328	Max right	45	14.2	−440	−12,760	8.3	−390	−11,310
328	Max right	45	38.5	320	9,280	46.5	390	11,310
329	Center	65	16.9	−410	−11,890	11.6	−400	−11,600
329	Center	65	43.9	330	9,570	46.8	520	15,080
330	Center	65	8.4	−370	−10,730	14.1	−260	−7,540
330	Center	65	39.5	370	10,730	33.8	250	7,250
331	Max left	65	15.5	−370	−10,730	7.6	−620	−17,980
331	Max left	65	40.4	320	9,280	44.5	420	12,180
332	Max left	65	8.5	−350	−10,150	9.5	−420	−12,180
332	Max left	65	44.8	360	10,440	46.8	410	11,890
333	Max right	65	8.9	−390	−11,310	9.3	−380	−11,020
333	Max right	65	39.5	250	7,250	45.3	350	10,150
334	Max right	65	7.2	−250	−7,250	10.0	−290	−8,410
334	Max right	65	39.0	240	6,960	44.6	260	7,540
335	Center	73	13.8	−180	−5,220	10.1	−310	−8,990
335	Center	73	56.4	150	4,350	35.0	220	6,380

*Time from zero time (zero time = time when first deflection is recorded on right leg).

**Stress = strain Y, where Y equals the stretch modulus of elasticity (29 million for gun steel).

The reaction force reported at the base of the mortar barrel was 79,200 pounds.

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APPENDIX A

Development of Equation for Frictional Force (F_f)
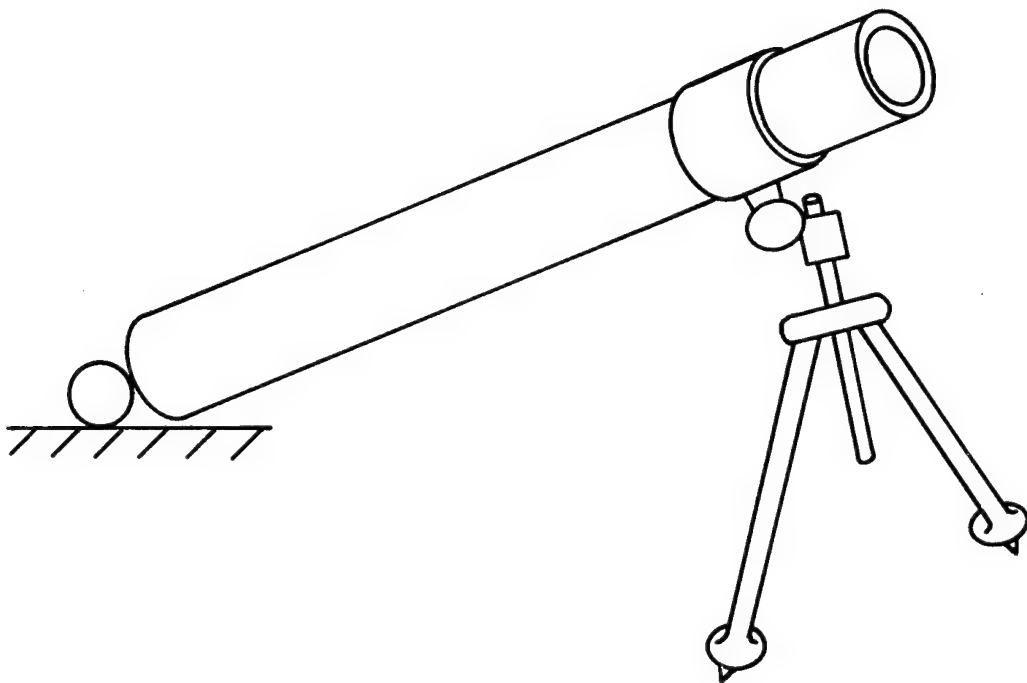
Fig 4   81 mm mortar
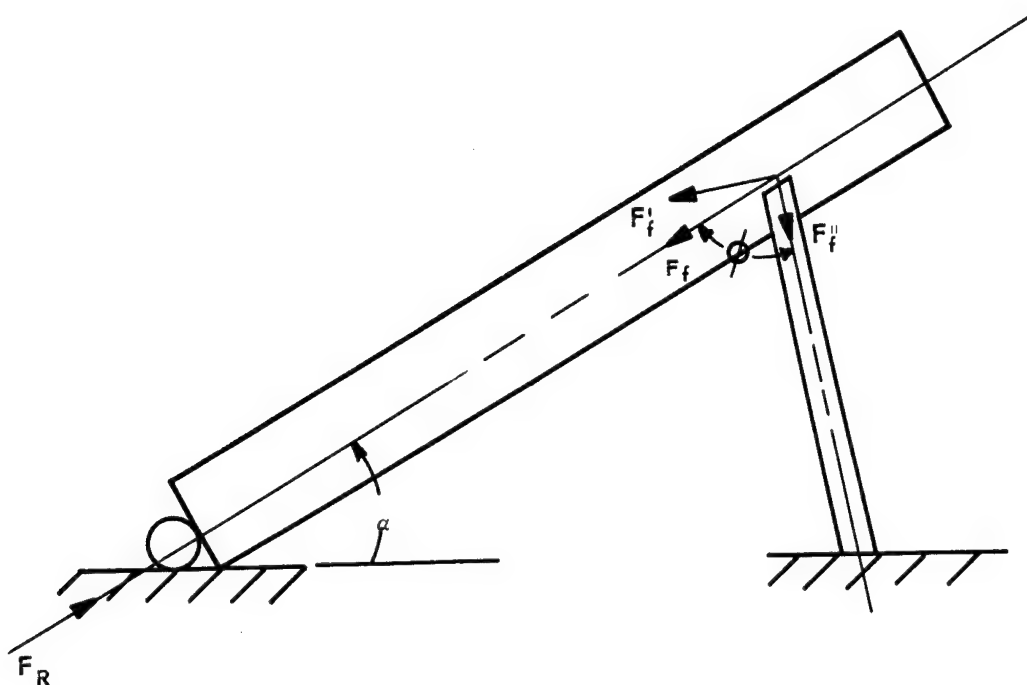
Fig 5   Schematic of force system

F_R = force of recoil

F_f = friction force

F_f' = bending component of friction force

F_f'' = compressive component of friction force along center line of bipod assembly, in plane of bipod

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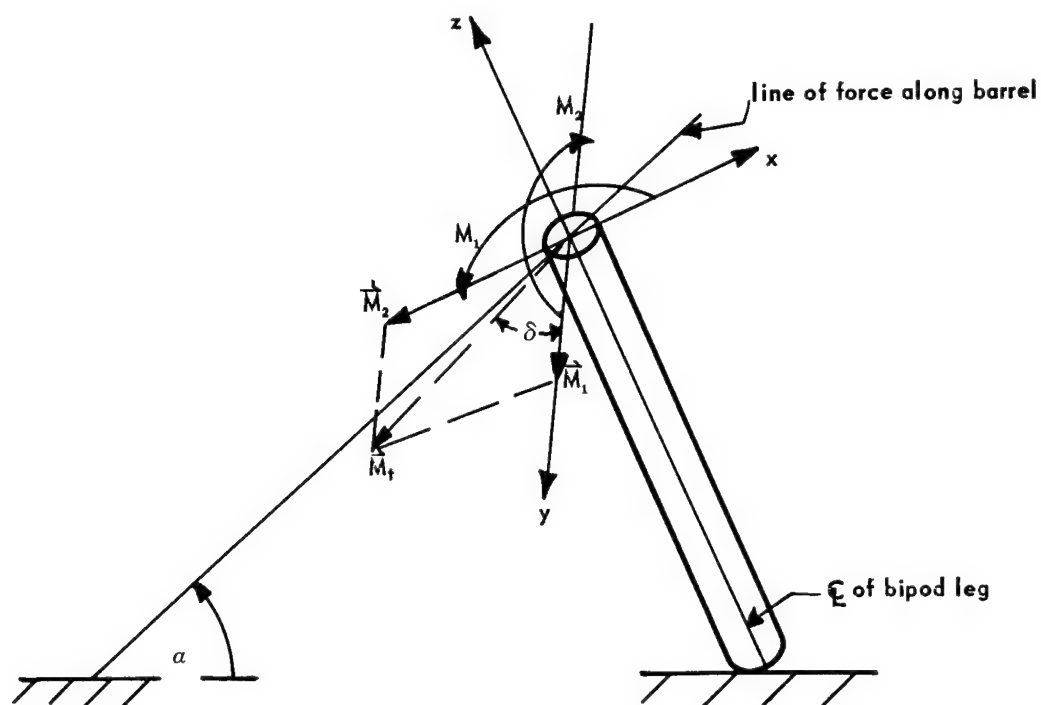
Fig 6   Moment diagram and edge view of bipod assembly

M_1 moment about plane of bipod assembly (x z plane)

M_2 moment in plane of bipod assembly   (y z plane)

$\vec{M_1}$ & $\vec{M_2}$ moment vectors

$\vec{M_t}$ resultant moment vector = $\vec{M_1} + \vec{M_2}$

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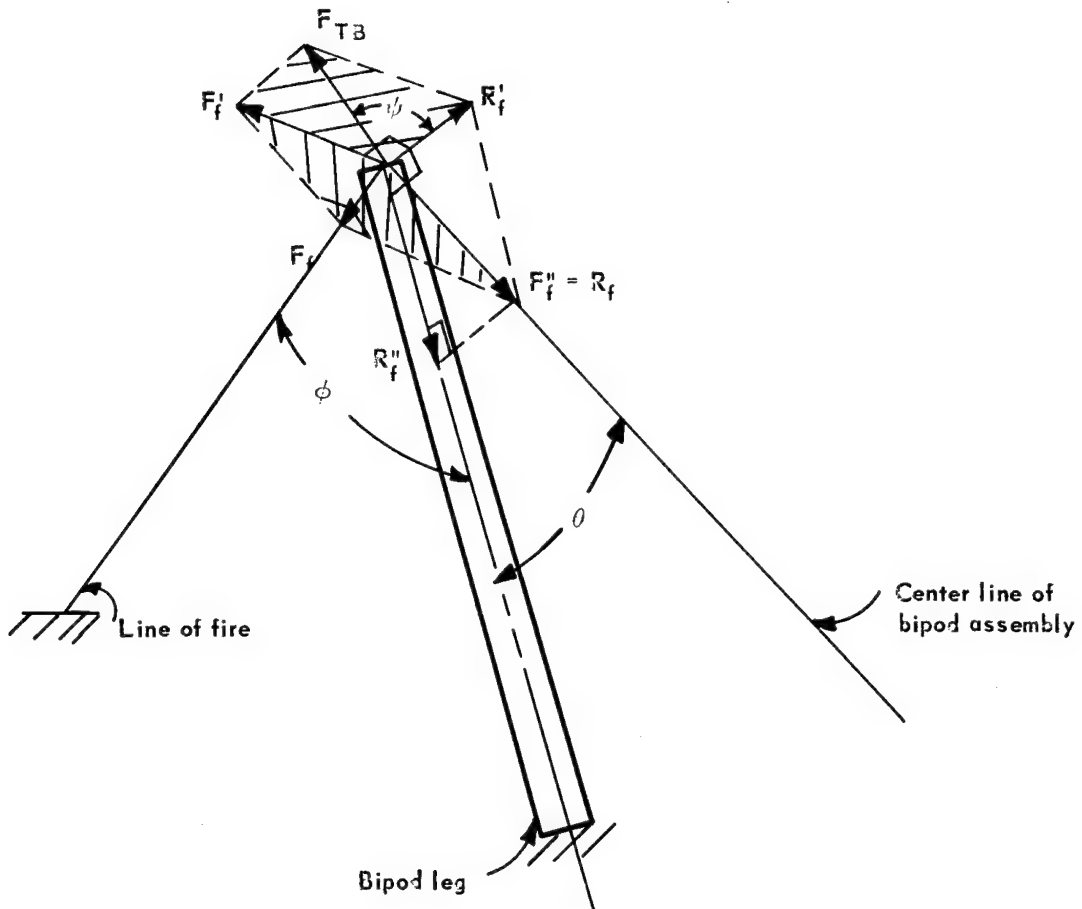
Fig 7   Force diagram

F_{TB} = resultant bending force

R_f = force acting along center line of bipod in the plane of the bipod assembly

R'_f = component of R_f or F''_f tending to bend bipod leg

R''_f = component of R_f compressing bipod leg

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Considering one leg of the bipod assembly to act as a cantilever thus bearing half of the applied load, we have

(a) Side view
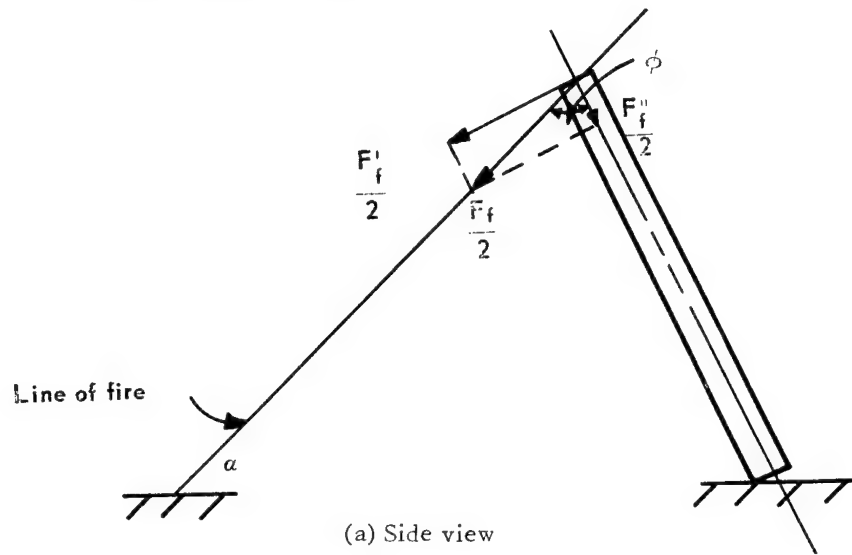
Each leg will bear a load of $\dfrac{F_f''}{2}$

(b) Front view

(c) Top view
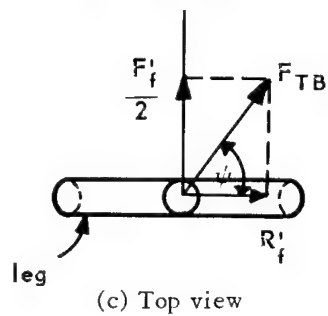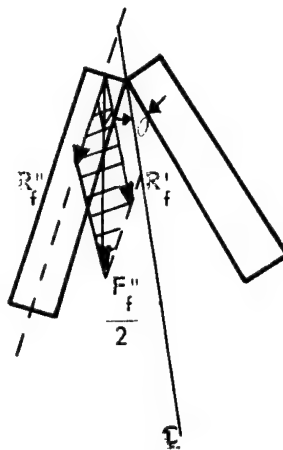
Fig 8   Bipod assembly

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From Figures 8 a, b, and c the following relations are readily determined

$$\left(\frac{F_f}{2}\right)^2 = \left(\frac{F_f^1}{2}\right)^2 + \left(\frac{F_f''}{2}\right)^2 \qquad (2a)$$

$$\left(\frac{F_f''}{2}\right)^2 = (R_f^1)^2 + (R_f'')^2 \qquad (2b)$$

$$(F_{TB})^2 = \left(\frac{F_f^1}{2}\right)^2 + (R_f^1)^2 \qquad (2c)$$

Adding the orthogonal equations (2a), (2b) and (2c) and making the following substitutions:

$$\frac{F_f''}{2} = F_{TB} \ \sin \psi \ \left[R_f^1 = \begin{cases} F_{TB} \cos \psi \\ \dfrac{F_f''}{2} \ \sin \theta \end{cases} \ R_f'' = \frac{F_f''}{2} \cos \theta \right] \qquad (2d)$$

we have after simplifying

$$\left(\frac{F_f}{2}\right)^2 = F_{TB}^2 + R_f''^2 \qquad (3)$$

The equations for bending and compressive stress for a cantilever are respectively,

$$\sigma_{TB} = \frac{M_T c}{I} = \frac{F_{TB} lc}{I} \qquad \text{(Bending)} \qquad (4a)$$

solving for F_{TB}

$$F_{TB} = \frac{\sigma_{TB} I}{lc}$$

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$$\sigma_c = \frac{R_f''}{A} \qquad \text{(compressive)} \qquad \text{(4b)}$$

solving for R_f''

$$R_f'' = \sigma_c \, A$$

l	length of bipod leg, in.
c	distance from neutral axis to extreme fiber, in.
I	moment of inertia of bipod leg, in.[4]
A	cross sectional area, in.[2]
F_{TB}	resultant force causing bending, lb
R_f''	compressive force applied to bipod leg, lb
F_f	frictional force, lb
σ_{TB}	tensile stress due to bending, psi
σ_c	stress due to compression, psi
σ_T	total stress $\sigma_{TB} + \sigma_c$, psi.

Substituting equations 4a and 4b into equation 3, we have,

$$F_f = 2\sqrt{\left(\frac{\sigma_{TB}I}{lc}\right)^2 + (\sigma_c A)^2} \qquad (5)$$

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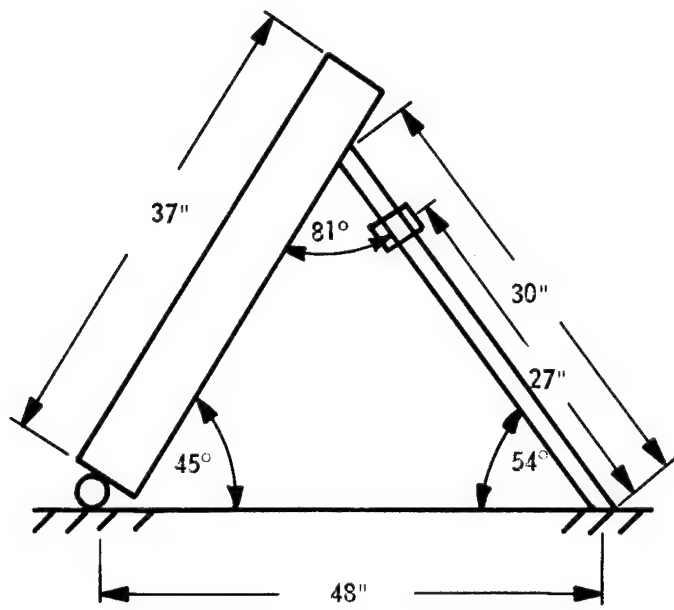
(a) Side view
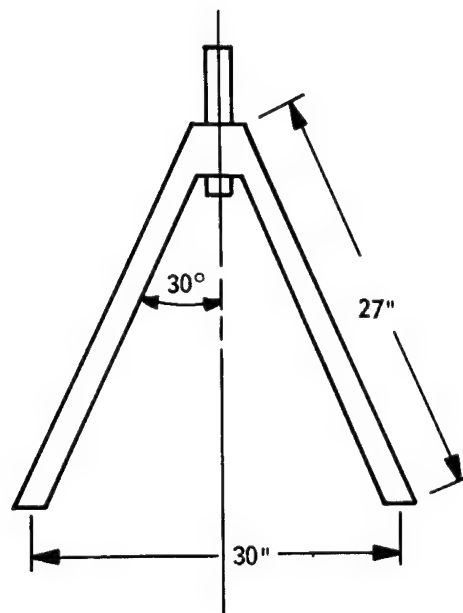
(b) Front view

Fig 9   Assumed geometry of bipod assembly during firing

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Using the following values, F_f is determined as a function of σ_{TB} & σ_c

$$I = \frac{\pi}{64} (D_o{}^4 - D_i^4) = \frac{\pi}{64} [(1.125)^4 - (.875)^4] = .0498 \text{ in.}^4$$

$l = 27$ in.

$$c = \frac{D_o}{2} = 0.563 \text{ in.}$$

$$A = \frac{\pi}{4} (D_o^2 - D_i^2) = 0.392 \text{ in.}^2$$

Substituting these values into equation (4), the result for the right bipod leg is,

$$F_f = \sqrt{(42.96 \times 10^{-6}) \sigma_{TB}{}^2 + (.616) \sigma_c{}^2} \tag{6}$$

Since the geometry and dimensions of the left and right legs differ, Equation 6 is applicable to the right bipod leg only. A similar expression for the left leg is presented in the discussion (Equation 1). Equations 1 and 6 differ only in the coefficients of σ_{TB} and σ_c.

APPENDIX B

The seven drawings contained in this appendix (Figs 10-16, pp 25-31) show how the components of the 81 mm mortar bipod would have to be changed, in design and dimensions, if they were to be manufactured from plastics. The principal changes shown are:

a. Relaxation of certain dimensional tolerances from .002 to .003 inch.

b. Deletion of instructions regarding surface finish.

c. If grease cups are to be used on bracket, allowance of sufficient clearance for bonding of cups.

d. A dimensional change of .0055 inch to leave sufficient clearance for bonding between case and housing tube connector.

e. Elimination of four drill holes in bottom of cylindrical portion of case, because plastic parts would be bonded.

f. Replacement of a .0615 + .005 inch hole in the case with a $\frac{1}{16}$ inch ream.

g. A dimensional change in the yoke ring to permit a clearance of at least .005 inch for bonding of the yoke ring.

All of the drawings included are modifications of the standard metal parts drawings, to incorporate the above-listed changes.

Oil grooves 1/16 R × 1/16 deep

.136(No. 29) drill
at assembly

5/16 + 1/64

2.05
−.02

1.998
(−.003)

1 21/32

3/32R

1 3/32

.6875
+.0030
(−.003)

1/2

.06 × 45° chamfer

1.84 diam. undercut

1 15/16 - 16N - 2

13/16

.498
−.003

1/8

5/64

3 R
32

5/64R

45/64

15/16

5/64

1 3/16

15/16

5/32

2 1/4

1/8

10°
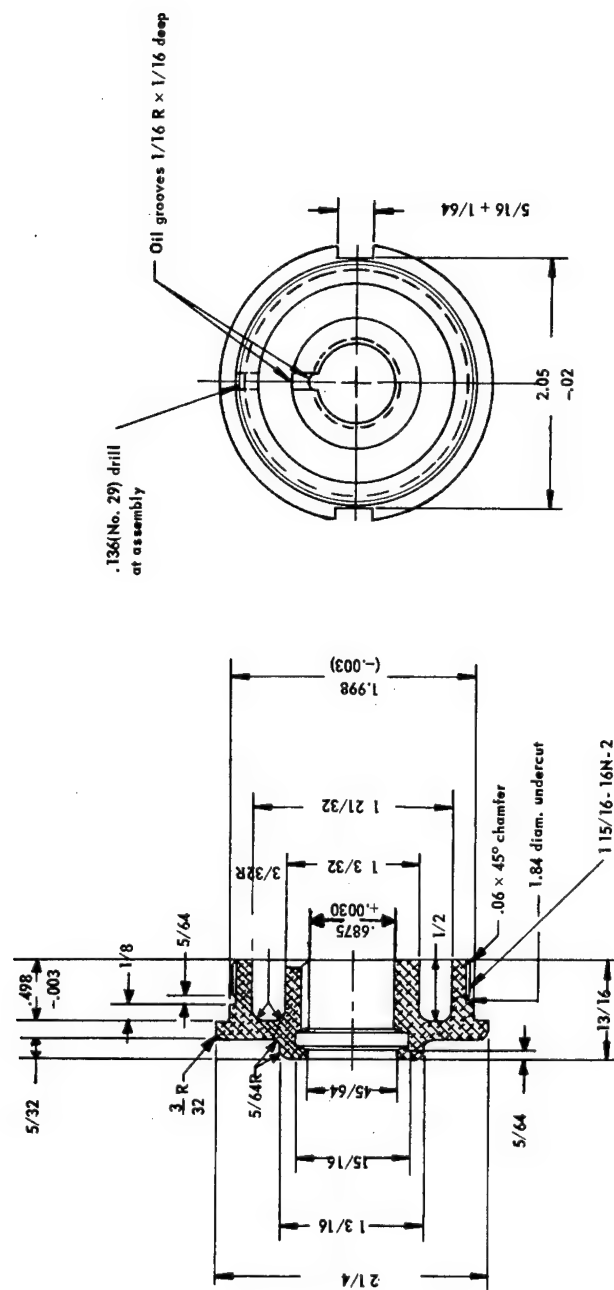
Fillets .01 max

Fig 10   Cover (Modification of Dwg 7236575)

25
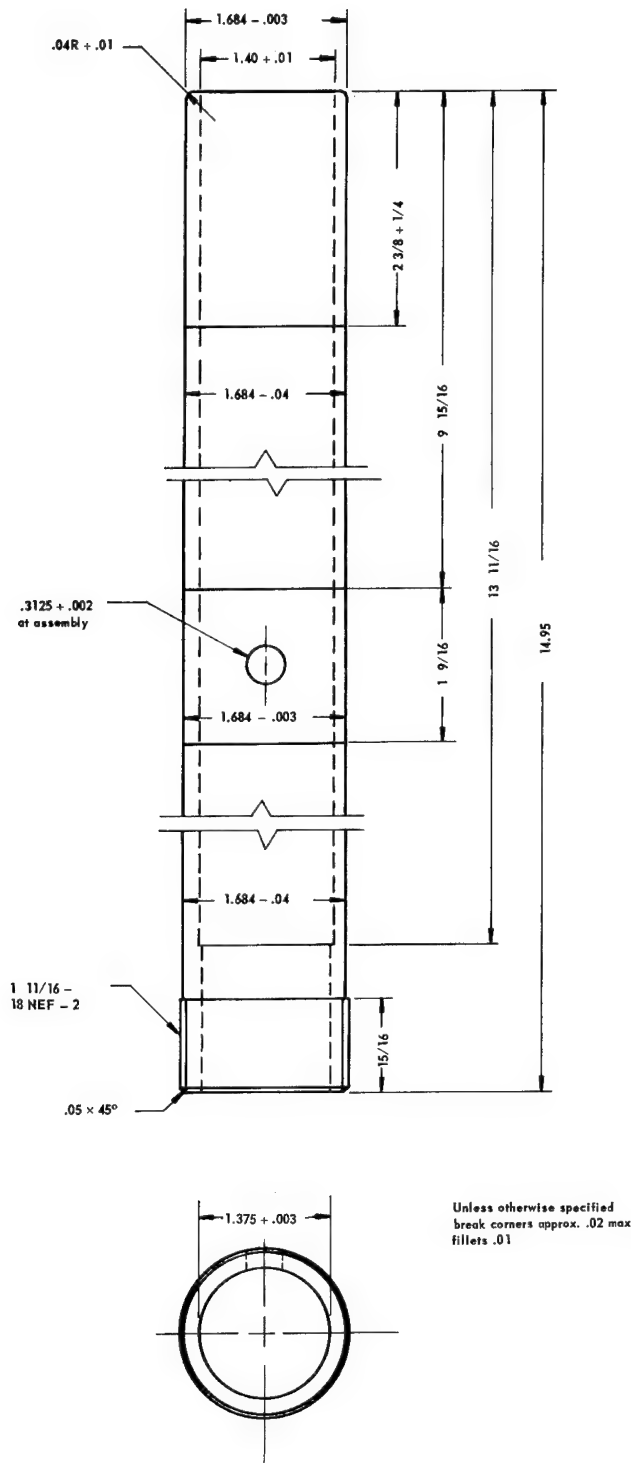
Fig 11   Tube (Modification of Dwg 7305150)

26
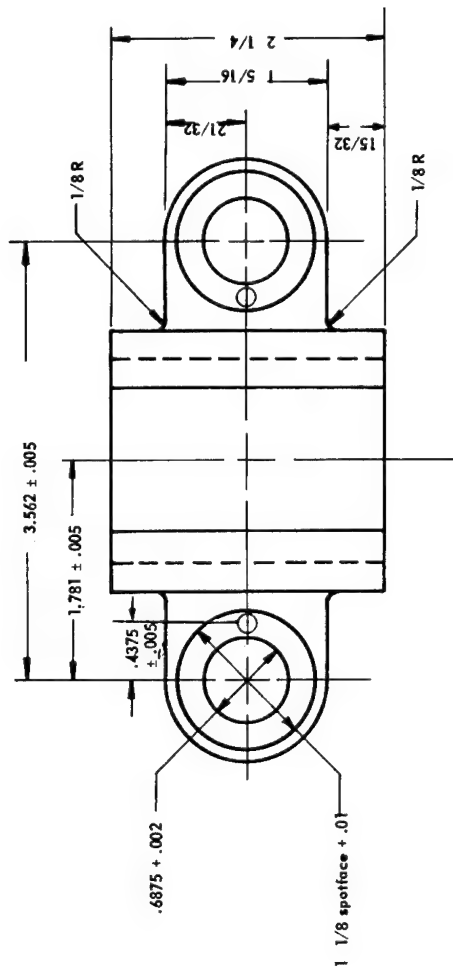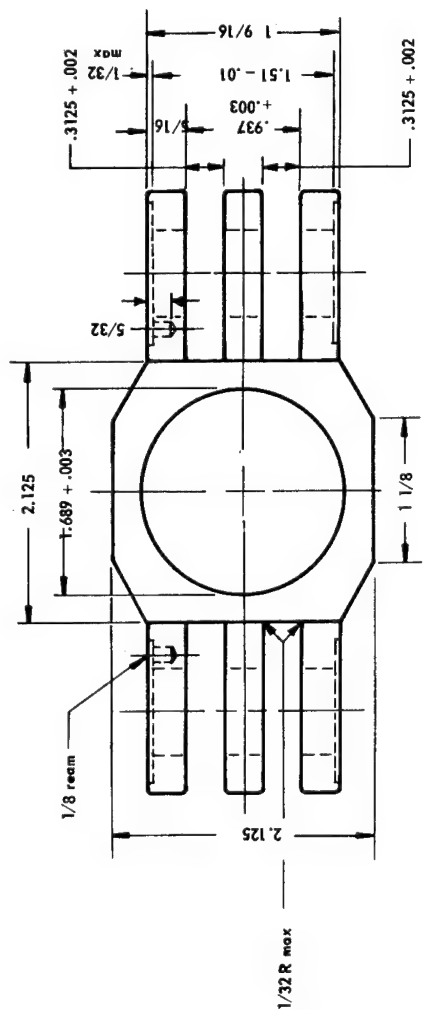
Fig 12   Housing tube connector (Modification of Dwg 7236033)

27
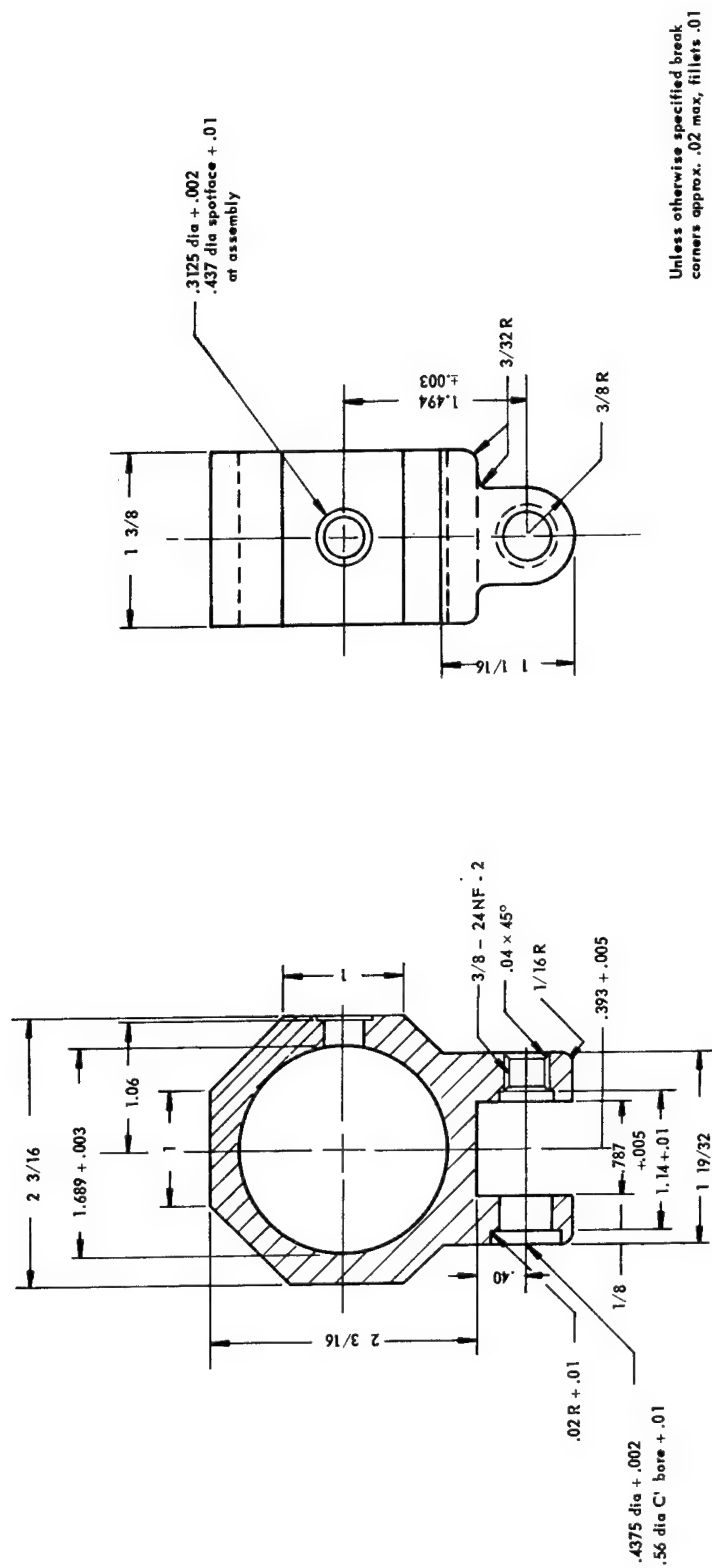
Fig 13   Bracket (Modification of Dwg 7305463)

Fig 14   Body (Modification of Dwg 7236576)

29

Fig 15   Case (Modification of Dwg 7235992)

Corners and fillets .115 max

.03 R + .01

.03 R + .01

4.00 − .01

3.563 + .003

3.807 − .003

Break corners .015 approx.
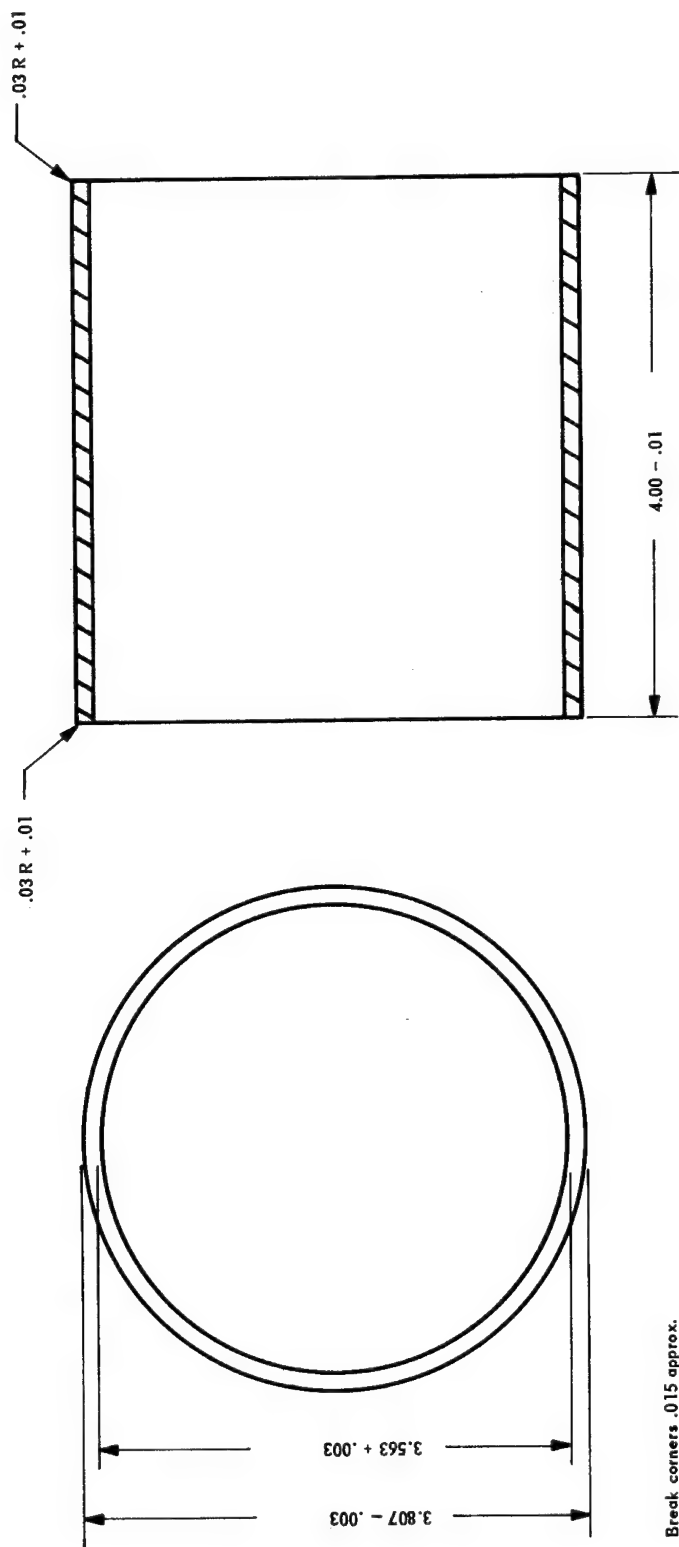
Fig 16   Yoke ring (Modification of Dwg 7305061)

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FEASIBILITY STUDY ON THE USE OF PLASTIC COMPONENTS FOR THE 81 MM MORTAR BIPOD ASSEMBLY
Merrill Eig

Picatinny Arsenal Technical Report 3030, February 1963, 39 pp, tables, figures. DA Proj 593-32-007, OMS 5010. 11.842. Unclassified Report

Strain data obtained during firing from gages positioned on the legs of the 81 mm mortar bipod assembly was reported to Picatinny Arsenal by Watervliet Arsenal. The forces exerted on the bipod assembly were determined by analyzing this data. Once this was accomplished the individual forces acting upon components could be calculated. A 45° elevation was used, with the bipod traverse assembly located in the central position.

(over)

1. Mortars, 81 mm – Materials
2. Epoxides – Applications
3. Glass reinforced plastics – Applications
I. Eig, Merrill

UNITERMS

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Mortar
Base
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Bipod
Eig, M.

The calculated maximum stresses were as follows: a bearing stress of 20,100 psi on the case; and a 2780 psi stress due to bending, and a compressive stress of 22,880 psi for the tube housing. The cover, body, and yoke ring were not significantly stressed. The connector showed a bending stress of 17,900 psi and a shear stress of 13,500 psi.

Rolled reinforced plastic tubing was suggested for fabrication of prototype components, as it can be easily machined to the desired shapes and has the required strength. It should be noted that the design strength of the tubing is marginal.

For end items, a glass-filled epoxy compression molding compound such as Scotchply 1100 would have the desired strength and other required physical properties.

The components were redesigned for fabrication from plastics. The recommended design changes are shown in an Appendix.

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